I. MANUAL PURPOSE (Revision 2)

To be used for selection, application into the system, power and cooling water estimation. This manual does not for designing centrifugal compressor and those parts.

II. MAIN COMPONENTS OF IN-LINE CENTRIFUGAL COMPRESSOR

Following figure (Fig.1) shows components of in-line centrifugal compressor.

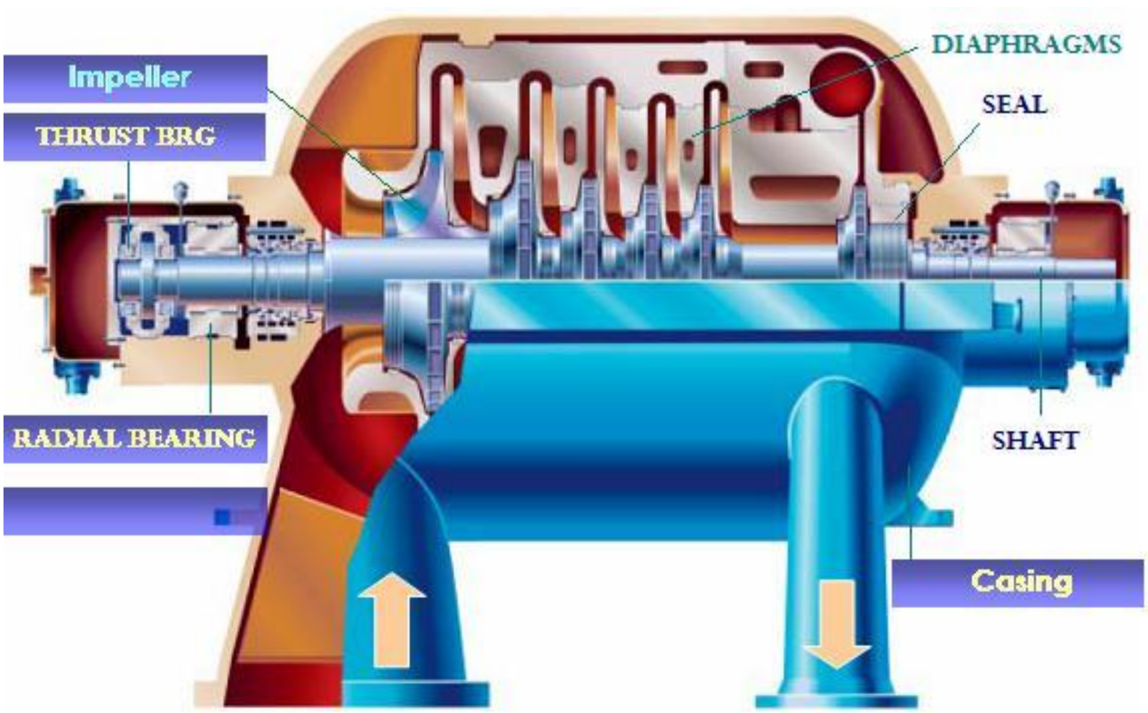


Fig.1. Typical horizontal split centrifugal compressor

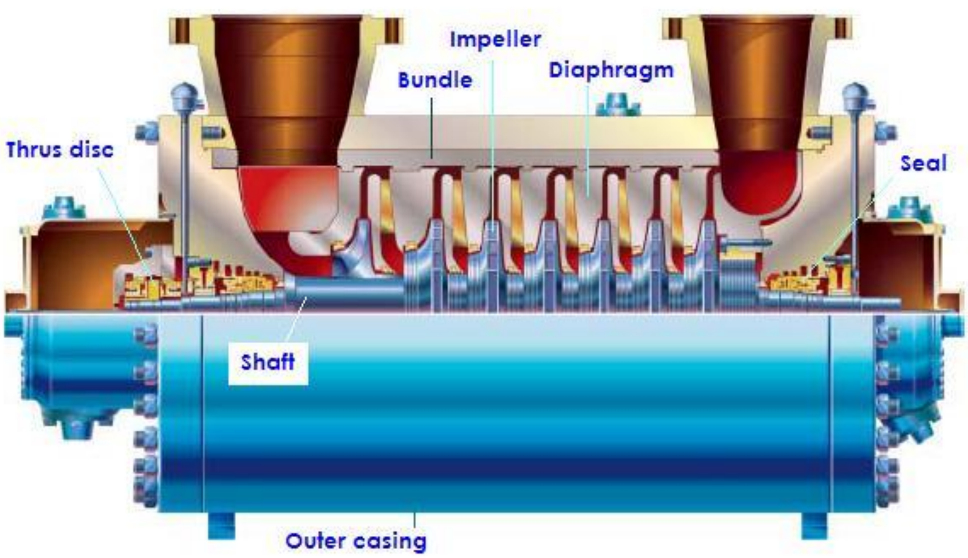


Fig. 2. Typical vertically split centrifugal compressor

Main components of centrifugal compressor are Casing, Shaft, Impellers, Bearings, Diaphragms and Seals.

II. SYMBOLS AND UNIT

Designation	**Symbol**	**Unit**
Pressure	p	bar A
Temperature	t	C
Absolute Temperature	T	K
Capacity (volume flow)	Q	m^3 / hr
Power	P	kW
Brake horse power	BHP	kW
Gas horse power	GHP	kW
Speed	N	RPM
Head	H	m
Gas Constant	R	kJ/kg.K
Molecular Mass	MW	kg/kg_{mole} $(=lb/lb_{mole})$
Mole	MM	kgmole (kgmole/h or kmol/h)
Density	DS	kg/m^3
Specific Gravity	SG	
Specific volume	v	m^3/kg
Specific Heat	Cp	kJ/kg.K
Mass Flowrate	G	kg / hr
Adiabatic Exponent	k	-
Polytropic Exponent	n	-
Compressibility Factor	Z	-
Efficiency	E	-
Gravity	g	m/s^2 (9.81)
Heat Capacity	MCp	kJ/kgmole
Enthalpy	h	kJ/kg
Enthalpy different	dh	kJ/kg
Entropy	s	kJ/kg.K
Impeller Diameter	D	mm
Tip speed (tangential)	U	m/s
Number of impeller	i	-
Mach Number	Ma	
Flow Coefficient	CQ	-
Head Coefficient	Y	-
Mechanical power loss	Pml	kW

Subscript

cr	Critical	i	Partial for gas, per impeller for impeller
red	Reduced	p	Polytropic
s	Suction	1, 2 etc.	Position
d	Discharge	I, II etc.	Stage or step
g	Gas (Horse Power)	n	Normal condition (0 C , 1.013 bar A)
STG	Stage or 1 casing	t	Total

III. UNIT CONVERSION

Designation	Unit to be converted	Factor	Unit to be used
Length	ft	304.8	mm
	inch	25.4	mm
Pressure	psi	0.06897	bar
	kg/cm^2 (at.)	0.981	bar
	atm.	1.013	bar
	Pa (Pascal)	10^{-5}	bar
Temperature	F (Fahrenheit)	(t-32) x (5/9)	C
	K (Kelvin)	T - 273	C
	R (Rankin)	(5/9)	K
Velocity	ft/s	0.3048	m/s
	ft/min (fpm)	0.00508	m/s
Volume flow	GPM (US)	0.227	m^3/hr
	Cfm	1.699	m^3/hr
Mass	lbm	0.4536	kg
Power	HP	0.7457	kW
Head	ft	0.3048	m
Enthalpy	kcal/kg	4.1868	kJ/kg
	BTU/lbm	2.326	kJ/kg
Gas constant	kcal/kg.K	4.1868	kJ/kg.K
Specific heat & Entropy	BTU/lbm.R	4.1868	kJ/kg.K
Specific mass or density	lbm/ft^3	16.0185	kg/m^3
Specific volume	ft^3 /lbm	0.06243	m^3/kg
Viscosity	$N.s/m^2$	1000	cP
	$lbf.s/ft^2$	47880.3	cP

Note : American Standard State at 1.013 bar A and 15.5 C. In volume common written as SCF.
Normal condition at 1.0132 bar A and 0 C. In volume common written as Nm^3

IV. OPERATING RANGE OF CENTRIFUGAL COMPRESSOR

Fig.3 presents operating range of Centrifugal Compressor based on suction flow and discharge pressure and fig. 4 presents operating range of centrifugal compressor compare with other type of compressor based on suction volume flow and speed.

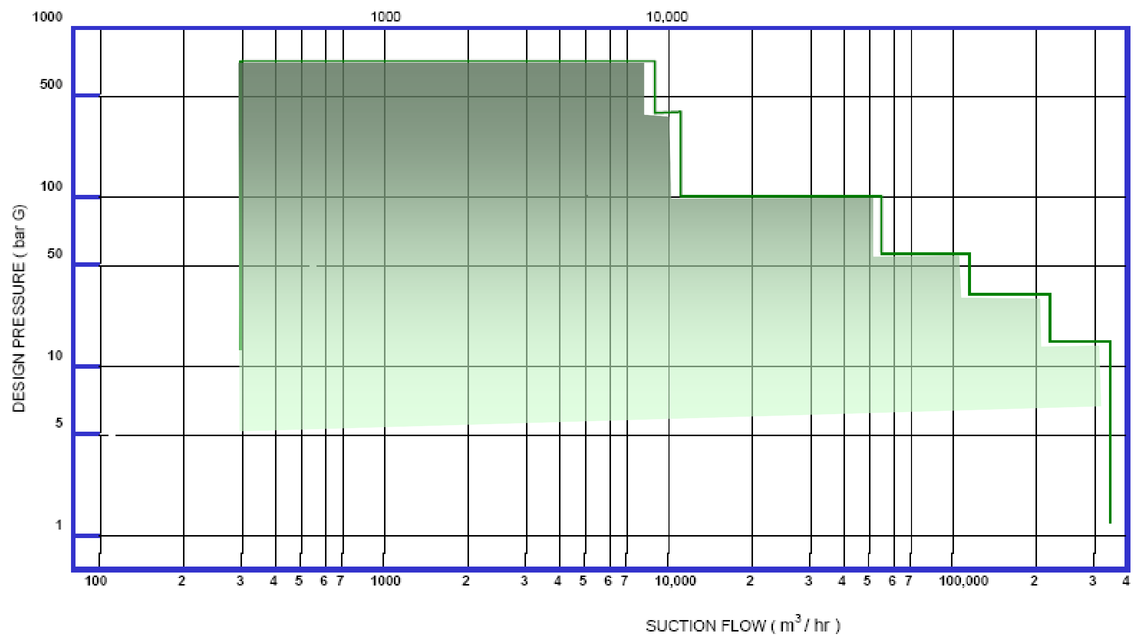


Fig. 3. Operating range of centrifugal compressor

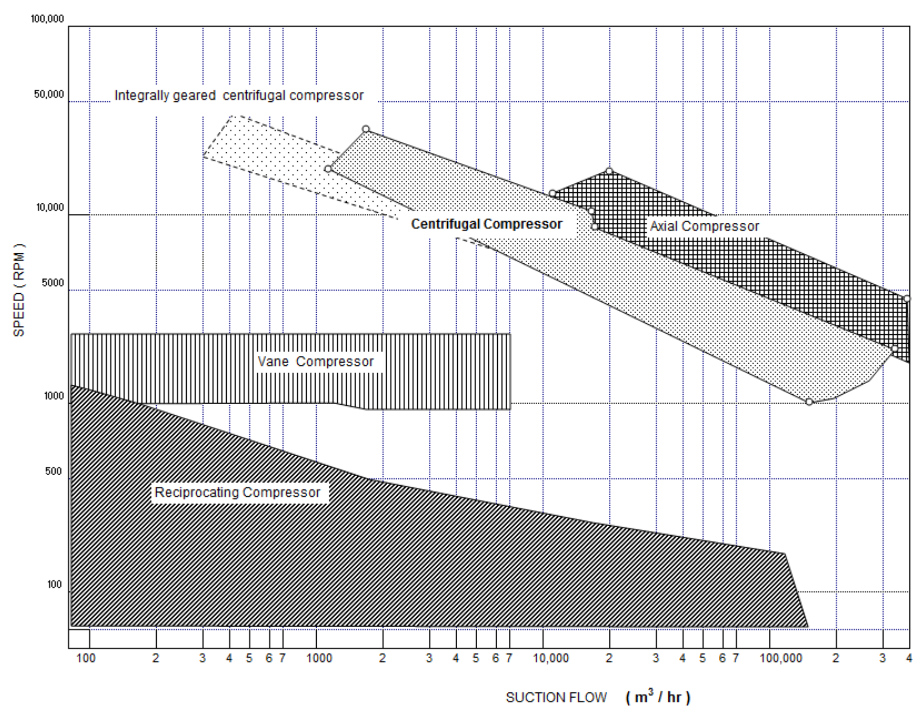


Fig. 4. Operating range of centrifugal compressor compare with other type of compressors

V. GAS COMPRESSION

Gases to be handled by compressor are both single component (pure gas) and mixed gas. This manual also describes physical properties of mixed gas.

In the next equations and calculations, gas is assumed as ideal gas but then corrected by correction factors and so ever is assumed equal to actual physical properties of the gas. By any reason, for some cases, compressed gas is also assumed as non ideal gas.

Compression process of gas in centrifugal compressor describes with Fig. 5.

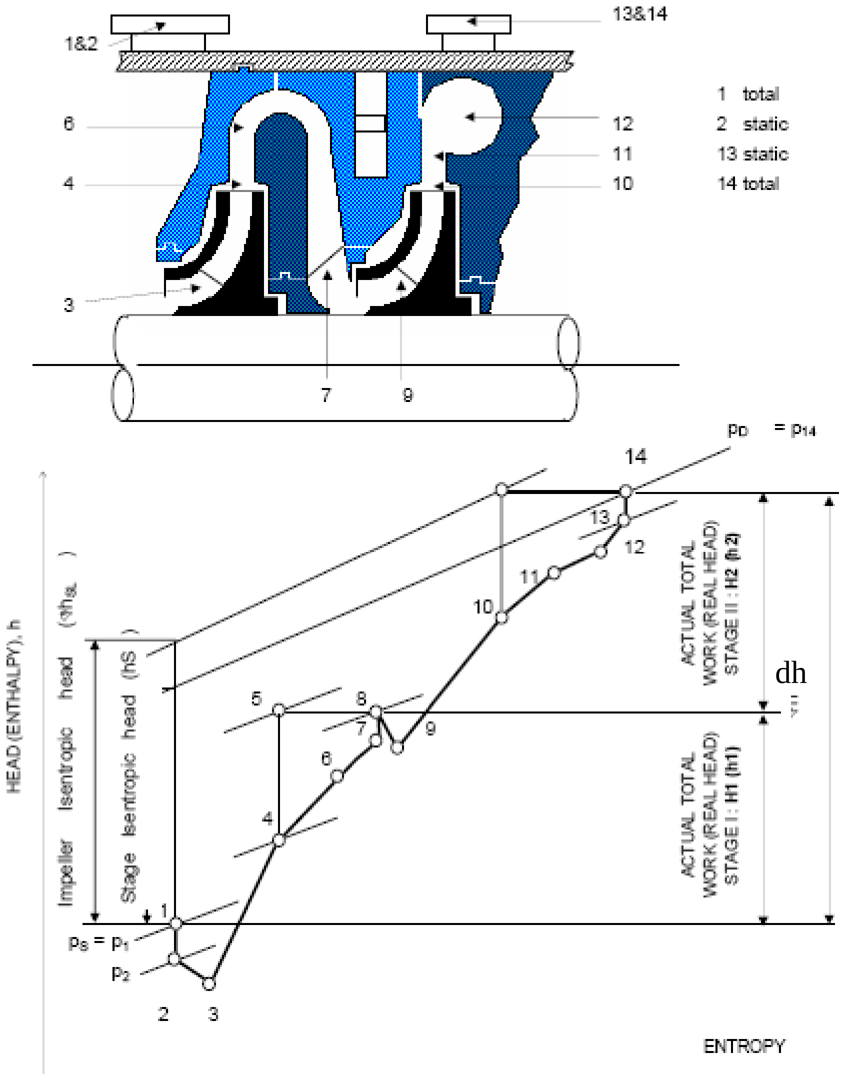
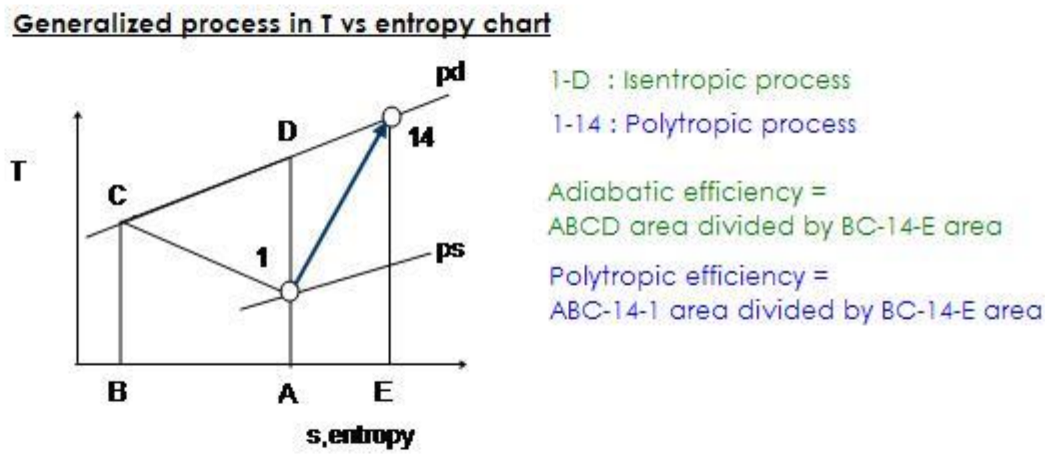


Fig. 5. Compression process of gas in centrifugal compressor

Gas compression process is presented in enthalpy versus entropy chart. Gas enter compressor through suction nozzle (1) at ps = p1 measured as total pressure and become p2 as static pressure in isentropic process. Gas goes to 1'st impeller eye (3) with little losses and then compressed to condition (4) and then in diffuser (6). Gas flows through vane (redirected) until condition (7) then come into 2'nd impeller eye (9). Next compression is in 2'nd impeller through condition (10) until (13). Gas goes out through discharge nozzle at condition (14) or at p2=pd. To determine adiabatic and polytropic efficiency, use the following chart and equations.



VI. INTERCOOLER

After compression, gas temperature will rise up but it is limited before entering to the next compression. Temperature limitation is depending to what sealing material to be used and gas properties. To decrease temperature before entering to the next compression, compressor needs intercooler.

Gas	Temperature limitation (C)
General gas	250 for labyrinth seal 180 for oil film seal or mechanical seal
Ammonia	160
Hydrocarbon	120
Freon	120
Chlorine	110
Acetylene	60

VII. AFTERCOOLER

Aftercooler is used when discharge gas temperature leaving compressor shall be decreased before entering to other equipment or system.

VIII. ANTISURGE CONTROL

Antisurge control shall be installed to centrifugal compressor because at low flow, compressor will surge. Antisurge control is instruments to detect pressure and flow at where compressor will surge. Antisurge control is completed with control valve to by pass discharge gas back to suction or vented to atmosphere. See Fig. 6.

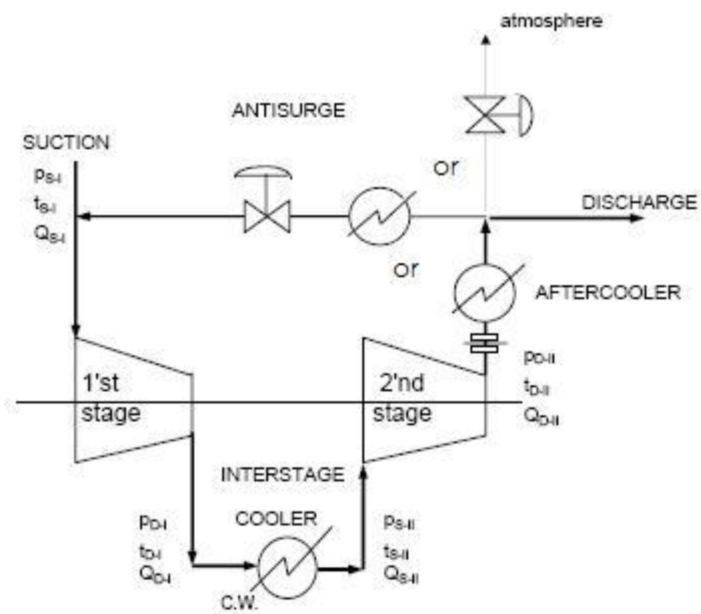


Fig. 6. Typical antisurge control for centrifugal compressor

IX. EQUATIONS

This manual uses the following simple equations. All symbols and unit are according to symbols and units described in chapter II.

Brake horse power

BHP = (GHP/EFp) + Pml (kW) (1) r2

Where (GHP/EFp) : **Horse Power up to impeller**, Pml : **Mechanical Losses** r2

Pml = Pml at bearing + Pml at seal = Pmlb + Pmls (kW) (2)

Pmlb = R_L $(0.001\ N)^2$ (kW), 2 journal brg. + 1 thrust brg

Pmls = R_S $(0.001\ N)^2$ (kW), 2 Oil Seal type

Pmls = 0.001 .R_D . N (kW), 2 Mechanical Contact Seal

R_L , R_S and R_D are factor depending to suction flow

Suction flow (m^3 / hr)	R_L	R_S	R_D
2500	0.13	0.07	0.54
10,000	0.45	0.24	1.1
50,000	2.9	1.55	3.2
300,000	10	5.4	6.5

Flow losses through labyrinth seal is between 1 s/d 5 %. Smaller compressor has bigger flow loss percentage.

Gas horse power

GHP = $\dfrac{G.H.g.10^{-6}}{3.6}$ (kW) (3) r2

Where G is mass flow = DSs.Qs (kg/h), DSs is density (kg/m3), Qs is vol. flow (m3/hr) (4)

See **Appendix B** for polytropic efficiency (EFp)

For perfect gas, **suction volume flow** is

Qs = $\dfrac{Ts.Qn.Zs}{269.69(ps.Zn)}$ and Qd = $\dfrac{Td.Qn.Zd}{269.69(pd.Zn)}$ (m^3/hr) (5)

Where Qn is volume flow at normal condition (0 C and 1.013 bar A)

DSs = $\dfrac{100(ps)}{Rs.Ts.Zs}$, DSn = $\dfrac{101.3}{273(R.Zn)}$ and DSd = $\dfrac{DSs.Ts.pd.Zs}{Td.ps.Zd}$ (kg/m^3) (6)

Hydrodynamic head (in polytropic process)

Hp = $\dfrac{1000(Zs)(R)(Ts)}{g}\{\dfrac{n}{n-1}\}\{(\dfrac{pd}{ps})^{(\frac{n-1}{n})}-1\}$ (m) (7)

Where (pd/ps) is compression ratio. Pd and ps are in absolute pressure (bar A) and

$$\frac{n}{n-1}=\frac{k}{k-1}(EFp)$$ (8)

R = Ro / MW (kJ/kg.K)

Ro = 8.314 (kJ/kgmole.K)

See **Appendix A** for R, MW, k and Z.

Note for Brake or shaft horse power r2

BHP = $\dfrac{GHP}{EFpt}$ (kW) when total polytropic efficiency (EF_{pt}) is known where (9) r2

bearing and seal losses are included in this total efficiency. If mechanical losses are calculated separately, than this EFpt is not necessary and **BHP = GHP/EFp+Pml** (see equation 1). Data of efficiency in this manual is EFp (Efficiency of impeller) r2

Discharge Temperature

Td = Ts . $(\dfrac{pd}{ps})^{(\frac{n-1}{n})}$ (10)

If discharge temperature is limited at Tdmax, then maximum pressure ratio become

$$(\frac{pd}{ps})_{MAX} = (\frac{Td\max}{Ts})^{(\frac{n}{n-1})} \quad (11)$$

Equations related to impeller geometry

Impeller tip speed,

$$U = \frac{3.14(D)(N)}{60{,}000} \quad \text{(m/s)} \quad (12)$$

Impeller tip Mach number ratio,

Mau = U / a, where a = $(1000.k.Z.R.T)^{0.5}$

Flow coefficient,

$$CQ = \frac{353.68(Qs)}{(U)(D^2)} \quad \text{in the range} \quad 0.01 \text{ up to } 0.15 \text{ see } \textbf{Appendix B} \quad (13)$$

Head coefficient,

$$Y = \frac{19.62(Hp)}{U^2} \quad (14)$$

Y values in the range 0.80 up to 1.1 for impeller with "backward leaning blades"
1.30 up to 1.45 ----- ,, --------- "90 degree exit blades "

X. PERFORMANCE CURVE

In general, centrifugal performance curve presented in head or discharge pressure against suction flow, see Fig. 7.

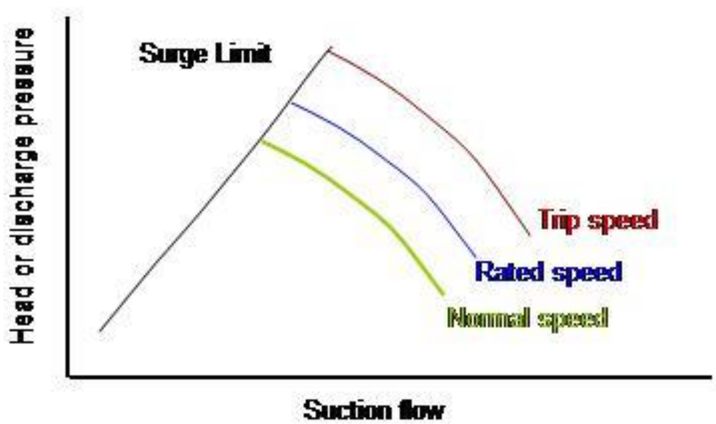


Fig. 7. Typical performance curve of centrifugal compressor (for variable speed driver)

APPENDIX A. GAS PROPERTIES

A.1. SINGLE GAS

The following table presents single gas properties. There are MW (molecular weight), k (adiabatic exponent), pcr (critical pressure), Tcr (critical temperature) and MCp (=MW x Cp).

Table 1. Pure Gas Properties

Gas or Vapor Name	Hydrocarbon Refer. Symbols	Chemical formula	MW (kg/kgmol)	k at 15.5 °C	Critical condition		MCp (kJ/kgmol.°K)		
					p_{CR} (bar A)	T_{CR} (°K)	at 0 °C	at 100 °C	at 197 °C
Acetylene	$C_2 =$	C_2H_2	26.04	1.24	62.4	309.4	42.16	48.16	53.17
Air (dry)		N_2+O_2	28.97	1.4	37.7	132.8	29.05	29.32	-
Ammonia		NH_3	17.03	1.31	112.8	406.1	34.65	37.93	-
Argon		Ar	39.94	1.66	48.6	151.1	20.79	20.79	20.79
Benzene		C_6H_6	78.11	1.12	49.2	562.8	74.18	103.52	-
Iso-Butane	iC_4	C_4H_{10}	58.12	1.1	36.5	408.3	89.75	116.89	141.88
n-Butane	nC_4	C_4H_{10}	58.12	1.09	38	425.6	93.03	117.92	141.04
Iso-Butylene	iC_4 _	C_4H_8	56.1	1.1	40	418.3	83.36	104.96	124.87
Butylene	nC_4 _	C_4H_8	56.1	1.11	40.2	420	83.4	105.06	-
Carbon Dioxide		CO_2	44.01	1.3	74	304.4	36.04	40.08	43.7
Carbon Monoxide		CO	28.01	1.4	35.2	134.4	29.1	29.31	29.63
Chlorine		Cl_2	70.91	1.36	77.2	417.2	35.29	35.53	35.9
Coke Oven Gas [1)]		-	10.71	1.35	28.1	109.4	31.95	34.21	-
n-Decane	nC_{10}	$C_{10}H_{22}$	142.28	1.03	22.1	619.4	218.35	280.41	-
Ethane	C_2	C_2H_6	30.07	1.19	48.8	305.6	49.49	62.14	74.37
Ethyl Alcohol		C_2H_5OH	46.07	1.13	63.9	516.7	69.92	81.97	-
Ethyl chloride		C_2H_4Cl	64.52	1.19	52.7	460.6	59.61	70.16	-
Ethylene	C_2 _	C_2H_4	28.05	1.24	51.2	283.3	40.9	51.11	60.55
Flue Gas [1)]		-	30	1.38	38.8	146.7	30.17	30.98	-
Helium		He	4	1.66	2.3	5	20.79	20.79	20.79
n-Heptane	nC_7	C_7H_{16}	100.2	1.05	27.4	540.6	161.2	202.74	239.8
n-Hexane	nC_6	C_6H_{14}	86.17	1.06	30.3	508.3	138.09	174.27	206.88
Hydrogen		H_2	2.02	1.41	13	33.3	28.67	29.03	29.25
Hydrogen Sulfide		H_2S	34.08	1.32	90	373.9	33.71	35.07	36.88
Methane	C_1	CH_4	16.04	1.31	46.4	191.1	34.5	40.13	44.64
Methyl Alcohol		CH_3OH	32.04	1.2	79.8	513.3	42.67	55.32	-
Methyl Chloride		CH_3Cl	50.49	1.2	66.7	416.7	45.6	49.82	-
Natural Gas [1)]		-	18.82	1.27	46.5	210.6	34.66	39.54	-
Nitrogen		N_2	28.02	1.4	33.9	126.7	29.1	29.31	29.46
n-Nonane	nC_9	C_9H_{20}	128.25	1.04	23.8	596.1	197.07	253.1	-
Iso-Pentane	iC_5	C_5H_{12}	72.15	1.08	33.3	461.1	112.09	145.56	-
n-Pentane	nC_5	C_5H_{12}	72.15	1.07	33.7	470.6	115.21	145.94	173.96
Pentylene	C_5 _	C_5H_{10}	70.13	1.08	40.4	474.4	102.11	130.37	-
n-Octane	nC_8	C_8H_{18}	114.22	1.05	25	569.4	176.17	226.17	-
Oxygen		O_2	32	1.4	50.3	154.4	29.17	29.92	30.78
Propane	C_3	C_3H_8	44.09	1.13	42.5	370	68.34	88.68	107.71
Propylene	C_3 _	C_3H_6	42.08	1.15	46.1	365.6	60.16	75.7	90.54
Blast Furnace Gas [1)]		-	29.6	1.39	-	-	29.97	30.64	-
Cat Cracker Gas [1)]		-	28.83	1.2	46.5	286.1	46.16	57.31	-
Sulphur Dioxide		SO_2	64.06	1.24	78.7	430.6	38.05	40	45.7
Water Vapor		H_2O	18.02	1.33	221.2	647.8	33.31	34.07	34.9

Note : For MCp, use linier interpolation to determine MCp at other temperature.

Gas constant (R), specific heat (Cp) and k

Gas constant R = $\dfrac{8.314}{MW}$ (A.1)

Specific heat Cp = $\dfrac{R.k}{k-1}$ (A.2)

Value of k is constant for dry gas, see table above.

Compressibility factor (Z)

Z determined by gas compressibility chart using reduction temperature (Tred) and pressure (pred) as the variables. Tred = T/Tcr and pred = p/pcr. See following Fig. 8.

Density of gas,

DS = $\dfrac{100(p)}{R.T.Z}$ (A.3)

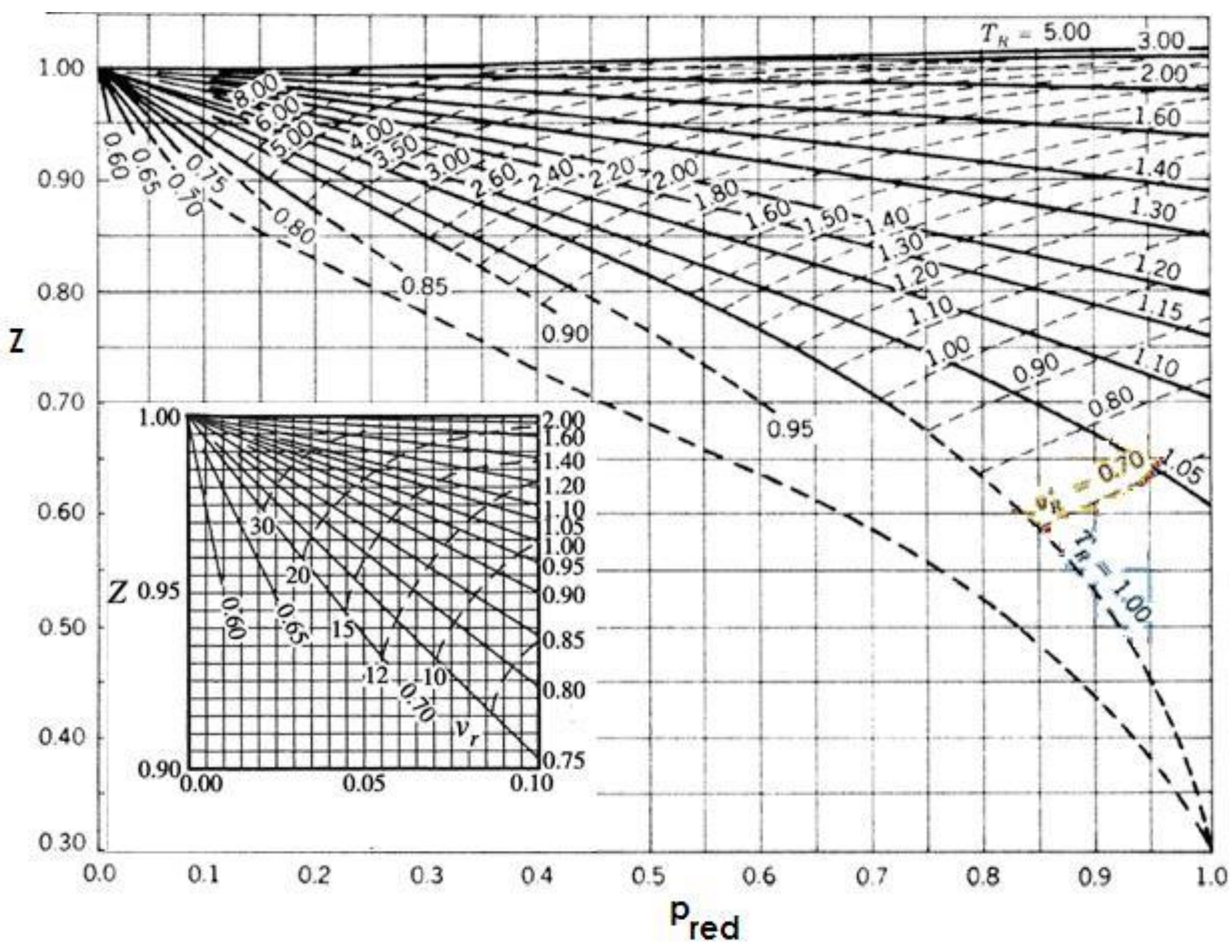


Fig. 8. Gas compressibility chart for pred < 1

Fig. 8 presents Z factor for pred = 1 and lower. For pred higher than 1 see Fig. 9.

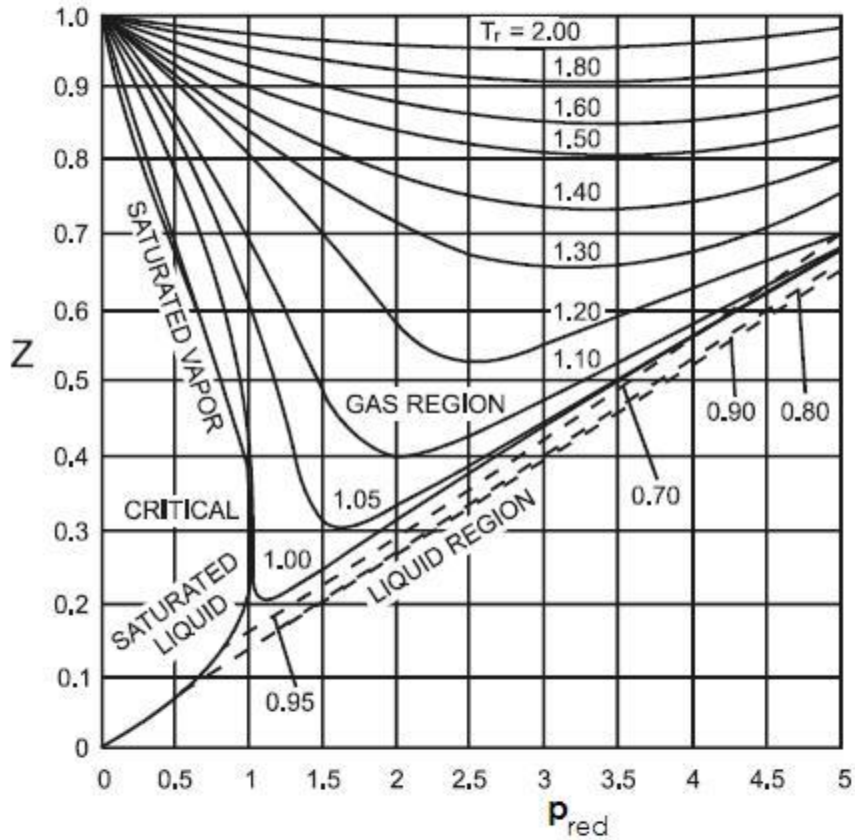


Fig. 9. Gas compressibility chart for pred higher than 1.

A.2. MIXED GAS

Gas constant (R), specific heat (Cp) and k of mixed gas

Subscript (i) indicates partial of pure gas.

$$MW = \sum^{i} \{0.01(\%Mi)(MWi)\} \tag{A.4}$$

Where %Mi is partial mole of each individual gas in %

$$\%\ Mi = \frac{100(MMi)}{\sum MMi} \tag{A.5}$$

Where MMi is molal mass of each gas in kgmole or mols

$$MMi = \frac{Mgi}{MWi} \tag{A.6}$$

Where Mgi is mass of each gas in kg

k = $\dfrac{\sum 0.01(MCpi)(\%Mi)}{\sum 0.01(MCpi)(\%Mi) - 8.314}$ (A.7)

Gas constant R = $\dfrac{8.314}{MW}$

Specific heat Cp = $\dfrac{R.k}{k-1}$

Compressibility factor (Z)

pcr = $\sum 0.01(\%Mi)(pcri)$ (A.8)

Tcr = $\sum 0.01(\%Mi)(Tcri)$ (A.9)

Z factor determined using Fig. 8 and 9 at above pcr and Tcr of mixed gas.

Density of mixed gas,

DS = $\dfrac{100(p)}{R.T.Z}$ (kg/m3, all unit shall be as listed in chapter II) (A.10)

A.3. WET GAS

Gas shall be dry in centrifugal compressor to prevent internal parts and impeller from erosion due to liquid particles. Gas condition shall be kept at little far from wet condition. Following table presents vapor pressure for some gases.

$p_{VAPOR} = A \times T^B$ bar A, and T at K

Gas name	A	B
Ethylene	1.64E-15	6.739
Ethane	1.02E-16	7.137
Propane	6.36E-19	7.702
Isobutane	2.42E-23	9.324
n-Butane	3.59E-25	9.984
n-Pentane	6.68E-30	11.647
n-Hexane	3.72E-35	13.568
n-Heptane	4.03E-31	11.794
n-Octane	5.85E-35	13.149
n-Decane	9.13E-42	15.533

Gas name	A	B
Carbon Dioxide	4.95E-19	8.141
HCl	4.26E-19	8.117
H_2S	1.29E-18	7.716
NH_3	1.23E-23	9.6
Cl_2	6.73E-22	8.86
SO_2	4.58E-27	10.816
Water (diatas 1 bar A)	6.71E-28	10.578

A.4. WET AIR

Following steps describes how to determine properties of wet air.

1. Relative humidity RH in %
2. Dry bulb temperature tdb in C and then Tdb = 273 + tdb in K
3. Atmospheric pressure patm at bar A

4. From psychometric chart, determine wet bulb temperature twb and Twb = 273 + twb
5. From H2O saturated pressure table, determine saturated pressure at twb, pg
6. Partial pressure of H2O pw = 0.01 (%RH)(pg)
7. Partial pressure of dry air pa = patm – pg
8. Mole fraction of dry air Xa = pa / patm
9. Mole fraction of H2O Xw = pw / patm
10. Molal mass of wet air MW = (MWdry air)(Xa) + (MW_{H2O})(Xw)
11. MCp of wet air MCp = (MCp dry air)(Xa) + (MCp H2O)(Xw)
12. Gas constant R = 8.314 / MW
13. k k = MCp / (MCp-8.314)
14. Density DS = 100.patm / (R.Tdb.Z)

Table 2. Saturated pressure of H2O

Temperature (C)	15	20	25	30	35	40	45	50
Sat.press. (barA)	0.01704	0.02337	0.03166	0.04241	0.05622	0.07375	0.0958	0.1233

Temperature (C)	55	60	65	70	75	80	90	100
Sat.press. (barA)	0.1574	0.1992	0.2501	0.3116	0.3855	0.4736	0.7011	1.0133

From % RH and tdb determine twb from following typical psychometric chart

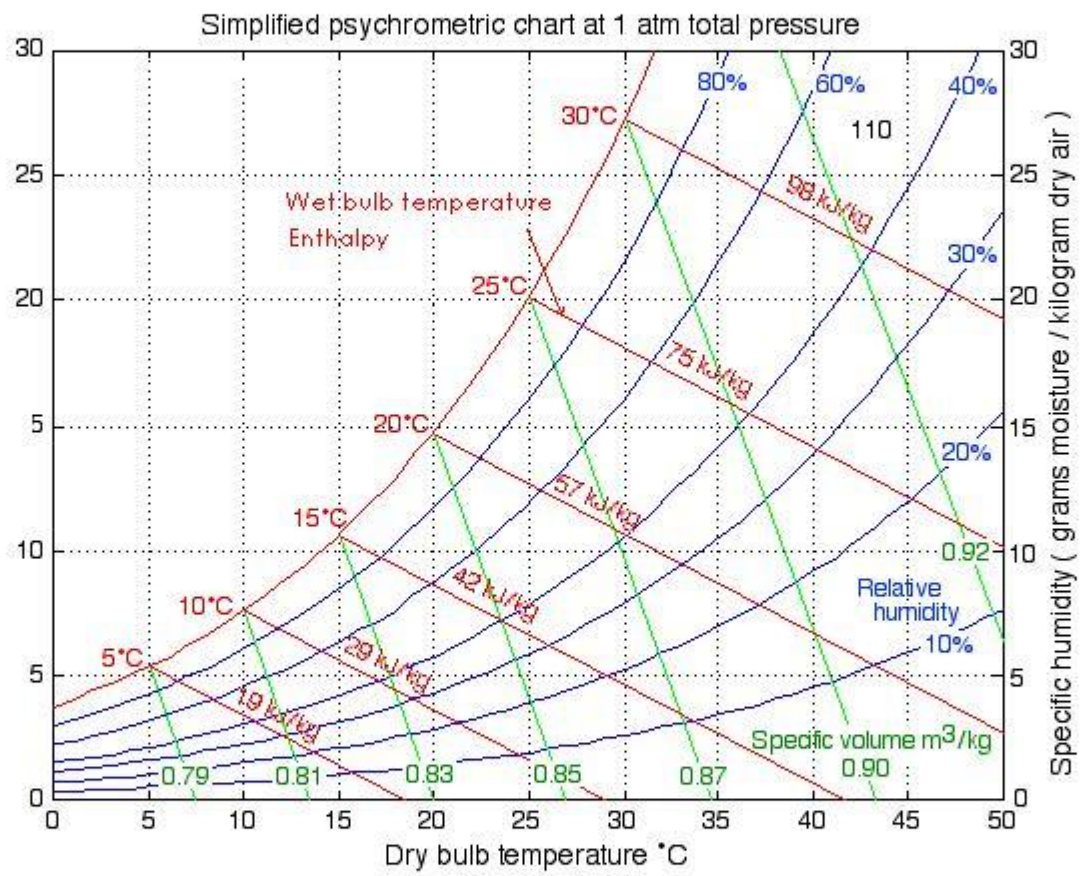


Fig. 10. Psychometric chart for air at 1 atm.

APPENDIX B. EFFICIENCY OF CENTRIFUGAL COMPRESSORS

A lot of parameter shall be considered in determining centrifugal compressor efficiency such as operating condition (flow, pressure, speed), impeller geometry and gas properties. In general, compressor manufacturer will offer compressor with efficiency as best as available after receiving user's specification. For preliminary estimation, use Fig. 11. In horizontal axis present actual flow which is not at standard condition or not at normal condition. Actual flow means suction flow.

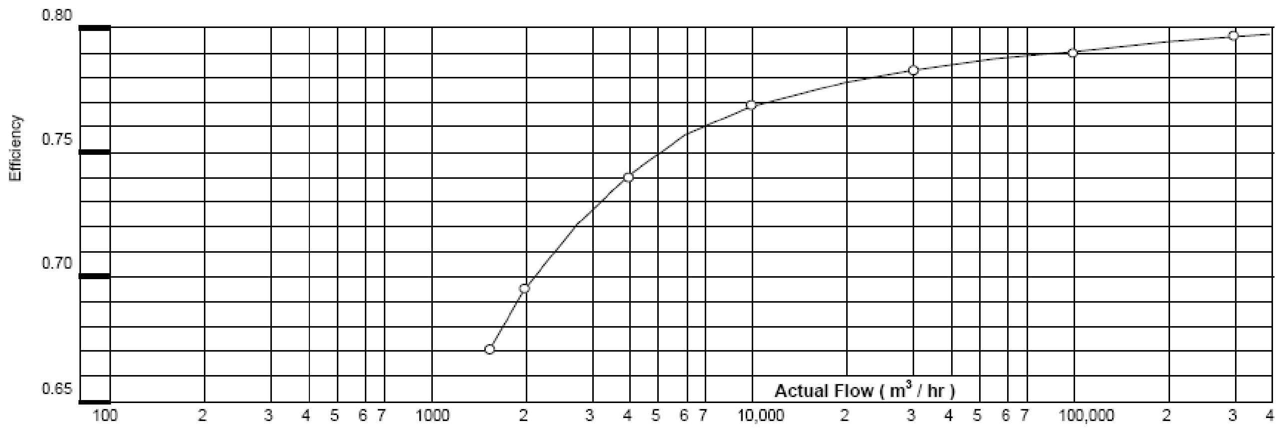
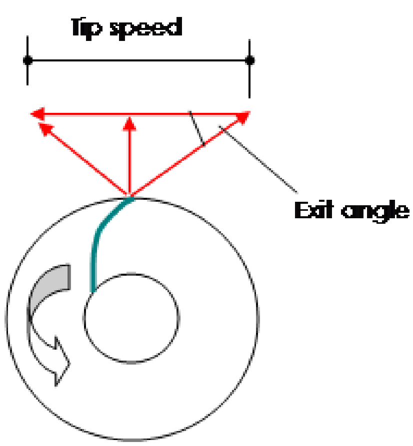
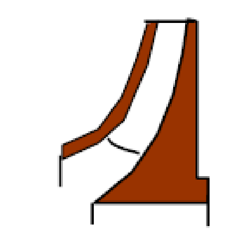


Fig. 11. Efficiency of centrifugal compressor for preliminary estimation.

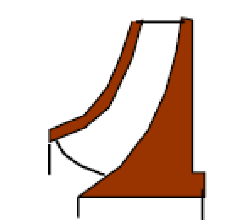
1. Impeller geometry and their characteristics

Following figure is simplified figure of impeller in relation with their performances. tip speed (U) and exit angle. Tip speed is limited due to material strength and sound velocity in compressed gas.

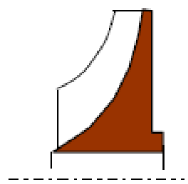




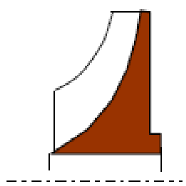
Shrouded Impeller
Radial inlet
Exit angle about 45 - 45 degree
Typical stage efficiency about 78 %
Max. polytropic head approx. 50 kJ/kg
Typical surge point 65 %
Relative max. flow coefficient 100%



Shrouded Impeller
Diagonal inlet
Exit angle about 50 degree
Typical stage efficiency about 83 %
Max. polytropic head approx. 55 kJ/kg
Typical surge point 50 %
Relative max. flow coefficient 150%



Unshrouded Impeller
Axial inlet
Exit angle about 60 degree
Typical stage efficiency about 81 %
Max. polytropic head approx. 85 kJ/kg
Typical surge point 70 %
Relative max. flow coefficient 150%



Unshrouded Impeller
Axial inlet
Exit angle about 90 degree
Typical stage efficiency about 77 %
Max. polytropic head approx. 105 kJ/kg
Typical surge point 75 %
Relative max. flow coefficient 200%

Fig. 12. Impeller geometry and their characteristic.

Shrouded impeller is equal with enclosed impeller where there is disc in the front integral with vanes or blades (casting, welded or riveted) .

Backward curve impeller is when exit angle smaller than 90 degree and radial curve impeller is when exit angle is equal to 90 degree.

To convert head from J/kg to m, divide J/kg unit by g (gravity in m/s^2). Example 55 kJ/kg = 55,000 J/kg = 55,000/9.81 = 5606.5 m

Tip speed is limited due to material strength and sound velocity in compressed gas.

There is criteria to determine tip speed such as the following table

MW	Average tip speed (m/s)
Below 35	310
Below 45	250
Below 65	200
Below 120	150

Even MW below 35 but gas contain corrosive matter or will be operated at low temperature below -50 C, maximum tip speed is 250 m/s.

Maximum tip speed shall not higher than sound velocity. For approaching, using Umax = 0.9 x a, where a is sound velocity = $(1000 \times k \times T \times Z \times R)^{0.5}$ or = $(8314 \times k \times T \times Z / MW)^{0.5}$. U= 3.14 x D x N /60,000, or D = 60,000 x U / (3.14 x N) or D = 60,000 X 0.9 x $(8314 \times k \times T \times Z / MW)^{0.5}$ / (3.14 x N) or D = 1568000 x $(kxTxZ/MW)^{0.5}$/ N. And "0.9" is factor for incorrect approaching for all assumption and calculation related to sound velocity and tip speed. This equation can be plot at several $(kxTxZ/MW)^{0.5}$, see Fig. 13 blue dot line. D and N pair under blue dot line is accepted because tip speed is lower than 0.9 time sound velocity.

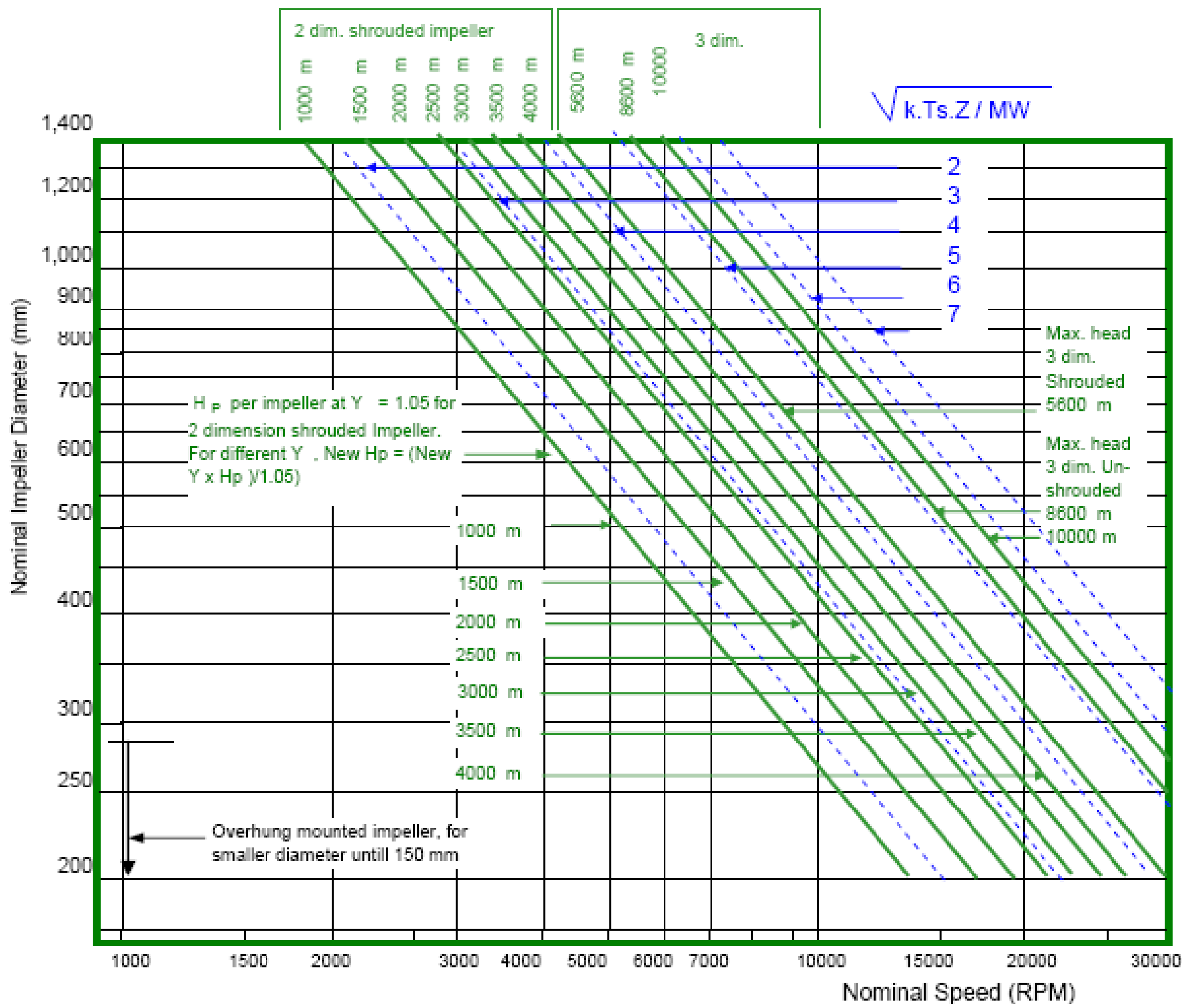


Fig. 13. Impeller performance in head

Example : Air at 30 C (303 K) and 1.013 bar A to be compressed 6 barA. Suction flow 2000 m^3 /hr. From table 1, k =1.4, pcr = 37.7 bar A, Tcr =132.8 K, MW =28.97. Determine pred = 1.013/37.7 = 0.027 and Tred = 303/132.8 = 2.28. From Fig. 8, Z near = 1. Ratio value of $(k.T.Z/MW)^{0.5}$ = 3.8. From Fig. 13, this line is near head per stage 4000 m line. At this line, if N=10,000 RPM can be determined diameter of 550 m and maximum tip speed U become = 3.14.D.N/60,000 = 3.14 x 550 x 10,000 / 60 ,000= 287 m/s.

2. Contribution of impeller geometry, speed and diameter for efficiency

Compressor size is designed according to suction flow of gas. Compressor size is includes impeller diameter, impeller width and number of impeller. Manufacturer usually indicates impeller diameter in each compressor model. The following Fig. 14 presents nominal size indicated by average impeller diameter. From suction flow can be seen approximately the nominal size and also speed in RPM.

Example : Continued from above example. At suction flow = 2000 m^3 /hr, draw vertical line until cross the range of each nominal size, provide nominal size 300 mm with impeller geometry “C”. Cross point between flow and maximum tip speed 287 m/s is at maximum about 20,000 RPM. In

C impeller geometry, at flow 2000 m^3 /hr speed in the range of 11,000 up to 20,000 RPM or tip speed from 200 up to 287 m/s.

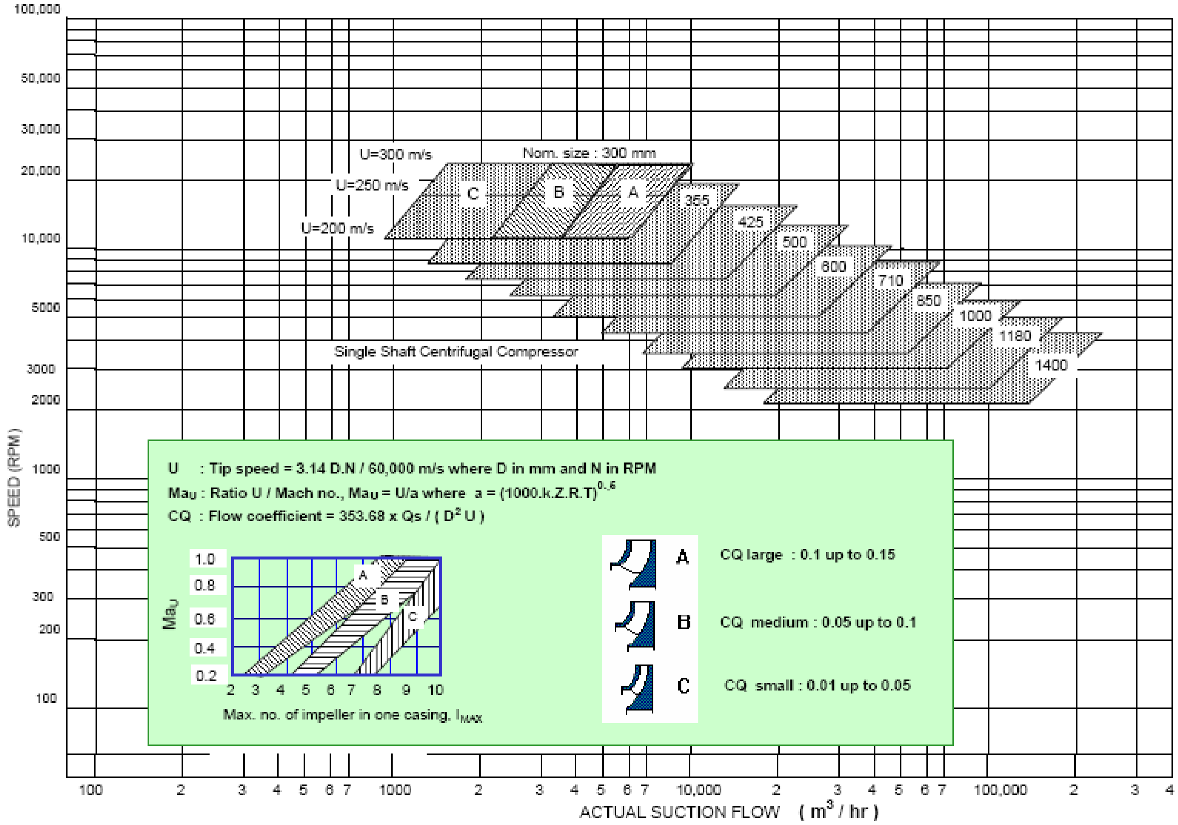


Fig. 13. Impeller performance in capacity

From rectangular bloc at the bottom of chart, geometry C impeller has flow coefficient CQ small = 0.01 up to 0.05.

Number of impeller in one casing is limited. Maximum number of impeller can be estimated from Fig. 13 as function of Mau and impeller geometry.

Example, continued from above example. If geometry C and nominal size 300 mm is selected, approximate diameter is 300 mm. And if tip speed is selected at 280 m/s, then Mau become = 280 / $(1000.k.R.T)^{0.5}$ = 280 /$\{1000 \times 1.4 \times (8.314/28.97) \times 303\}^{0.5}$ = 0.8. From Fig. 13 maximum number of impeller Imax = 10.

Number of impeller will decrease if there are additional nozzle is installed such as for inter cooler, admission or extraction. Each 1 nozzle will reduce 1 impeller.

Instead of preliminary efficiency that determined from Fig. 11, efficiency of compressor can also be determined after preliminary diameter and tip speed is selected. Fig. 14 shows efficiency as function of head coefficient and flow coefficient.

Example, continued from above example. D = 300 mm, U = 280 m/s have been selected at above example. N = 17,800 RPM. Assumed impeller is backward 2 dimension shrouded impeller.

Calculate flow coefficient CQ = 353.68 x Q / (U x D^2) = 353.68 x 2,000 / (280 x 300^2) = 0.028
Determine efficiency correction factor from Fig. 14. Co = 0.915

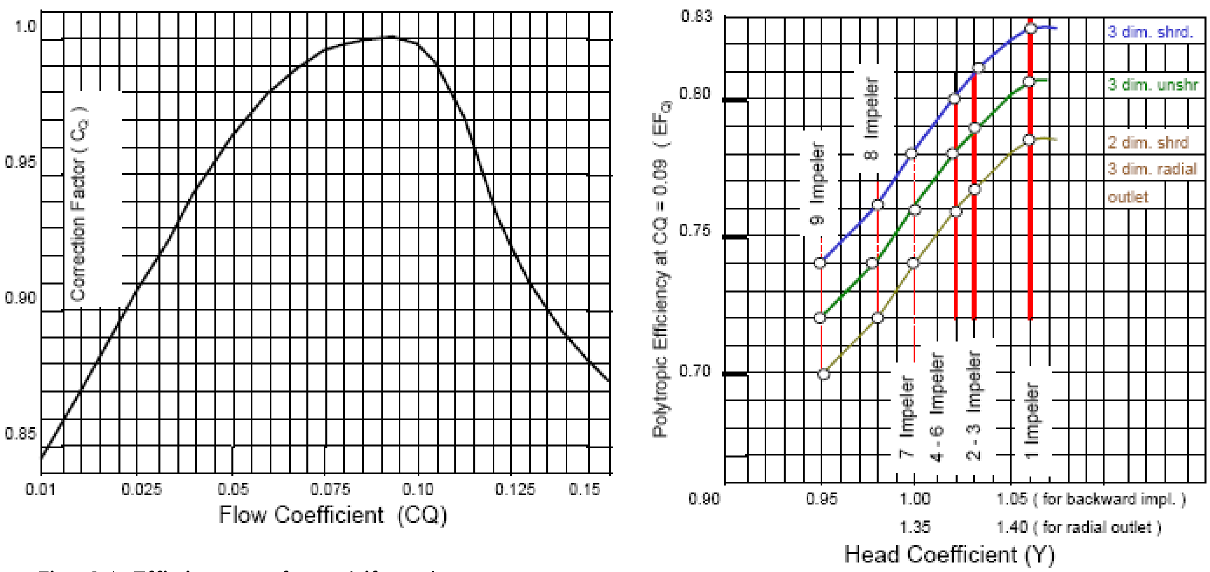


Fig. 14. Efficiency of centrifugal compressor

To determine head coefficient, preliminary efficiency shall be determine from Fig. 11, EFp= 0.695.

n/(n-1) = EFp. k/(k-1) = 0.695 x 1.4 /(1.4-1) = 2.432

assume maximum temperature tdmax = 200 C, Td1max = 473 K

pressure ratio max. at section 1 = $(\frac{pd}{ps})_{MAX}$ = $(\frac{Td\max}{Ts})^{(\frac{n}{n-1})}$ = 2.95

extract air to 1'st intercooler at pressure pd1 = 2.95 x 1.013 = 2.99 barA

and then entering to next compression at 40 C, Ts2=313 K

by the same way and assumed pressure drop is neglected aross intercooler,

pressure ratio at section 2, pd2/ps2 = 6 / 2.99 = 2

Hp1 = $\frac{1000(Zs)(R)(Ts)}{g}\{\frac{n}{n-1}\}\{(\frac{pd}{ps})^{(\frac{n-1}{n})}-1\}$ = 12,260 (m)

Plot again N=15,000 and D=300 mm on Fig. 12. Cross point is at head per stage = 3250 m

Number of impeller at section 1, I-1= 12,260/3,250 = 3.77 , take I-1 =4

Hp2 = 10,600 m

Number of impeller at section 2, I-2= 10,600/3,250 = 3.26 , take I-2 = 4

Total impeller = 4 + 4 =8 acceptable because Imax = 10 minus for nozzle 2 = 8

From Fig. 14 with number of impeller = 8 and 2 dim. shrouded impeller, uncorrected efficiency = 0.72. Corrected efficiency become

EFp = 0.72 x Co = 0.72 x 0.915 = 0.66 or 66 % . This result is smaller than first assumption = 0.695 or 69.5 %

APPENDIX C. CALCULATION SHEET AND SYSTEM (AS ATTACHMENTS)

Typical calculation sheets as ***attachment 1*** of this article are presented in excel file, named "Cal_sheet_c_comp.xls" which are included :

1. Pure gas properties
2. Mixed gas properties
3. Wet air properties
4. Power absorption and Cooling water required
   a. Without considering detail of impeller
   b. Considering detail of impeller

Typical system around centrifugal compressor as attachment 2 are presented in PDF file, named "Sys_c_comp,pdf" which are included :

1. Intercooler, aftercooler and antisurge system
2. Lube oil system
3. Compressor control
4. Sealing system